

Development of the performance of an alpha-type heat engine by using elbow-bend transposed-fluids heat exchanger as a heater and a cooler

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ABSTRACT

In this work, elbow-bend heat exchangers were suggested to be used as a heater and a cooler in an alpha-type Stirling engine. Elbow-bend heat exchanger is a bank of tubes arranged in a quadrant either in line or staggered with different normal and parallel pitches. Eight of such heat exchangers having different dimensions were tested experimentally for steady flow (in a previous work by the same authors). The experimental results were correlated for heat transfer and pressure drop. In the present work, an alpha-Stirling engine with twin parallel cylinders on a common crankcase was suggested to use elbow-bend heat exchangers as a heater and a cooler. In the heater, the flue gases flow inside the tubes and the working gas fluctuates about the heater tubes. In the cooler, the coolant flows inside the cooler tubes and the gas flows about the cooler tubes. A computer program in the form of a spread sheet was prepared to solve numerically the engine cycle in the vision of Schmidt theory. Upon calculations, the most suitable stroke/bore ratio, phase angle and speed were found out for nitrogen as a working gas. In a comparison among the proposed engine and practical ones by the literature, it was found that; the proposed engine delivers about 13% more power per cc per ΔT than those by the literature at high thermal efficiency level.

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1. Introduction

The conventional alpha-Stirling engine consists of twin cylinders. The heating and cooling processes can be achieved through the walls of the cylinders as shown in Fig. 1a. Another arrangement for heat transfer could be carried out by using a number of tubes that are bent in the form of a circular-quadrant as shown in Fig. 1b, where; the gas flows through the tubes and it can be heated by the flow of a heating fluid about the heater tubes and it can be cooled by the flow of a coolant about the cooler tubes. In this section, some of tubular heat exchangers that were used in Stirling machines will be reviewed as an introductory justification for the present work.

The SOLO 161 engine shown in Fig. 2 is an alpha engine, single acting, twin cylinders disposed in V-shape and the gas is helium, [1]. The heater has two headers; one for the gas coming from the expansion space and the other is for the gas coming from the regenerator. It was heated by natural gas at 740 °C. The cooler is a shell-and-tube type.

A technical feasibility study for a moderate temperature differential Stirling engine using nitrogen as a working gas which is heated by synthetic-oil having a temperature of about 280 °C

was performed by [2]. The engine is rated at 10 kW/700 rpm. A 3D unsteady CFD simulation for the tubular heater shown in Fig. 3 was done. Whereas; the working fluid fluctuates in a double-pass from expansion space to the regenerator. The cross-flow of synthetic-oil heats the gas during its flow through the heater housing and then it is in turn heated up by means of parabolic trough solar collectors or evacuated tube solar collectors.

A laboratory Stirling engine was tested at Lithuania Energy Institute by [3]. The engine has two parallel cylinders and it uses nitrogen as a working gas. The engine was heated directly by an electric current as the walls of a tubular heater are resistor heating elements as shown in Fig. 4. The cooler is a water-cooled one. The power loss from this engine due to hydraulic losses is about 15% of the indicated power.

The practical SM5 engine consist mainly of two cylinder connected by a cooler, a regenerator, and a heater, [4–6]. The engine has two volumes in the same cylinder separated by a displacer piston. The gas is heated up during its fluctuation from expansion space to the regenerator through a hair-pin heater shown in Fig. 5. The gas flows in a curved pass which increases the hydraulic losses and reduces the delivered power.

The biomass fired Stirling engine is considered to be the best technical and economical solution for a power range from 5 to 100 kW [7]. The heater of such engine consists of a number of

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Nomenclature

A	surface area (m ²)
a	cross section area (m ²)
cc	cubic centimeter
D	cylinder bore (m)
d	diameter (m)
f	friction factor
h	heat transfer coefficient (W/m ² K)
i	wire mesh (Pores/in.)
L	length (m)
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
N	speed (rpm)
NTU	number of transfer units
Nu	Nusselt number
P	power (W)
Pr	Prandtl number
p	pressure (N/m ²)
Δp	pressure drop (N/m ²)
Q	heat transfer rate (W)
R	specific gas constant (J/kg K)
Re	Reynolds number
S	stroke (m)
St	Stanton number
T	temperature (K)
ΔT	temperature difference between heat source and sink (K)
t	time (s)
U	overall heat transfer coefficient (W/m ² K)
V	volume (m ³)
ΔV	volume difference (m ³)
W	work (J)

Greek symbols

α	phase angle (rad)
ε	regenerator effectiveness
ϕ	crank angle (rad)
η	efficiency
μ	viscosity (kg/m s)
v	velocity (m/s)
ξ	temp. ratio = T_C/T_E
ρ	density (kg/m ³)
ψ	porosity

Subscripts

C	compression space
ch	charging
cl	clearance
E	expansion space
f	working fluid
g	flue gases
H	heater
hyd	hydraulic
i	inner conditions
K	cooler
max	maximum
min	minimum
o	outer conditions
R	regenerator
t	total
th	thermal
w	wire of the regenerator
wt	cooling water

curved tubes as shown in Fig. 6; whereas; the gas fluctuated through the heater tubes and the flue gases flow about the heater tubes.

Under consideration of solar energy in Turkey, a V-type Stirling engine having two heaters, double displacer and one power piston was designed by [8]. The two heaters increase the dead volume of the engine and reduce the delivered power. A prototype engine was tested using an electrical heating system. The engine still overcomes several problems; balancing, leakage, heat transmission and dead volume. If these problems are solved, it can be used as a stationary plant.

From the previous brief review and up to authors experiences, the elbow-bend transposed fluid heat exchangers have not been used in Stirling machines yet. Referring to Fig. 7, two photographs

of the suggested heat exchanger are presented. Eight ones of such heat exchanger were manufactured and tested experimentally in a steady flow conditions. The specifications of the eight heat exchangers are presented in Fig. 8. Experimental results were correlated by [9] as follows:

$$\overline{Nu}_f = A \times Re_f^B \quad (1)$$

$$f = C_1 \times Re_f^2 + C_2 \times Re_f + C_3 \quad (2)$$

where the constants; A , B , C_1 , C_2 and C_3 are given in [9].

In the present work, the elbow-bend heat exchanger will be introduced as a heater and a cooler in a twin-piston alpha-Stirling engine. Referring to Fig. 9, the working gas flows past the tube bank of the elbow-bend heater and cooler. In the heater, flue gas or heating fluid flows inside the tubes while the gas fluctuates

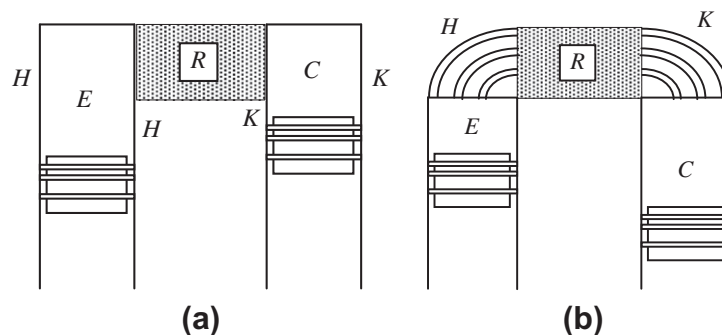


Fig. 1. (a) Alpha Stirling engine with the heating and cooling are through the cylinder walls. (b) Alpha Stirling engine with curved-tubes heater and cooler.

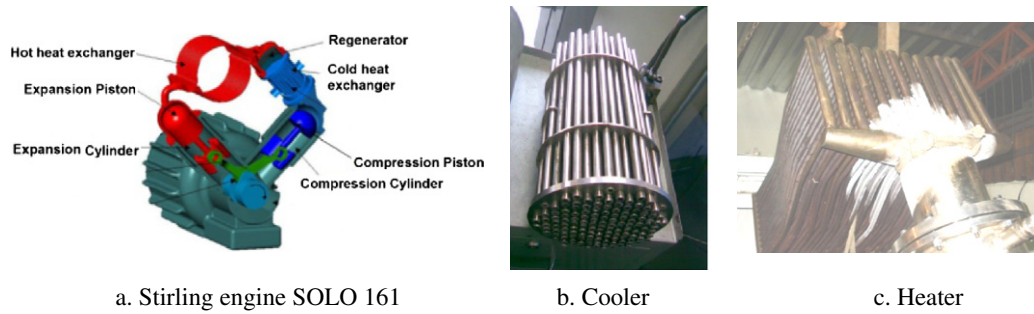


Fig. 2. SOLO 161 engine, shell-and-tube cooler and heater, after [1].

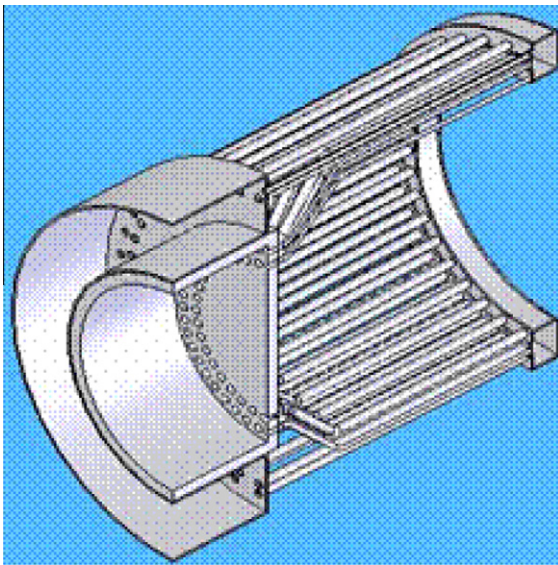


Fig. 3. Tubular heater for moderate temperature differential Stirling engine, after [2].

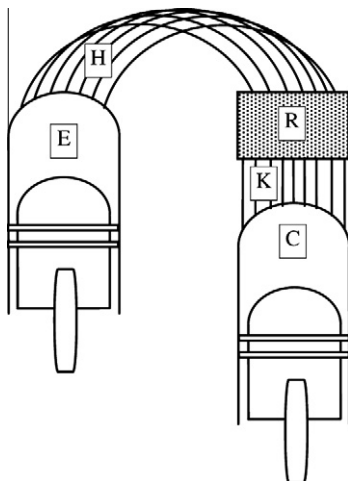


Fig. 4. Scheme of a laboratory Stirling engine in Lithuania Energy Institute, after [3].

about the heater tubes. In the cooler, the coolant water flows inside the tubes and the gas fluctuates about the cooler tubes. The selection of this type of heat exchanger is referred to its ease in manufacturing, its long life time, quite cheap, compactness, reliability, suitable for alpha and gamma engines and it has somehow low hydraulic losses as the gas flow is an external flow while the flow of both heating and cooling fluids are internal one.

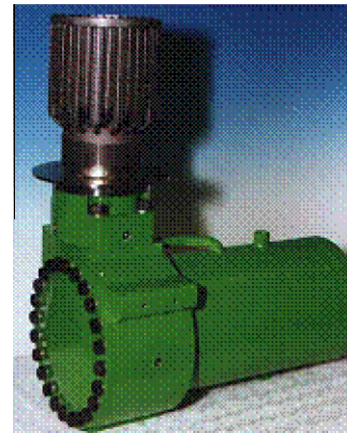


Fig. 5. SM5 Stirling engine, after [4].

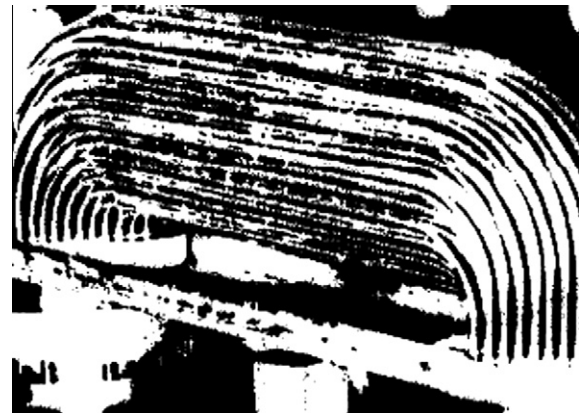


Fig. 6. Heater tubes of the test biomass Stirling engine, after [7].

2. Engine

The present work deals with an alpha engine having twin cylinders of 100 mm bore. The heater and cooler were located in the corners over both expansion and compression spaces. A wire-net regenerator was situated between the heater and the cooler as shown in Fig. 9. In the vision of Schmidt theory [10,11], a harmonic variation of both expansion and compression spaces is to be assumed, thus;

$$V_E = (\pi/8)D^2S_E(1 - \cos \phi) \quad (3)$$

$$V_C = (\pi/8)D^2S_C(1 - \cos(\phi - \alpha)) \quad (4)$$

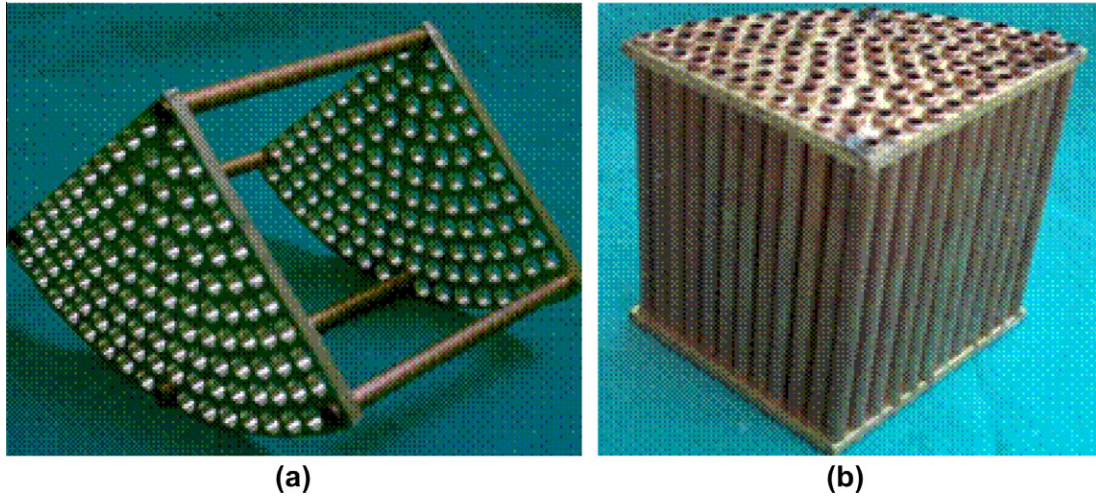


Fig. 7. (a) Elbow-bend heat exchanger during its assembly adjustment. (b) Elbow-bend heat exchanger after its assembly.

The total volume of the engine work space is as follows:

$$V_t = V_E + V_C + V_H + V_K + V_R + V_{cl,E} + V_{cl,C} \quad (5)$$

The intermediate temperature of the regenerator T_R is;

$$T_R = (T_E - T_C) / \ln(T_E / T_C) \quad (6)$$

The mass of the gas is;

$$m_{ch} = P_{ch} V_{t,max} / RT_{ch} \\ = p(V_E + V_H + V_{cl,E}) / RT_E + p(V_R) / RT_R + p(V_C + V_K + V_{cl,C}) / RT_C \quad (7)$$

The instantaneous Schmidt pressure is:

$$p = \frac{m_{ch} RT_E}{[(V_E + V_H + V_{cl,E}) + [\ln(1/\xi)/(1-\xi)](V_R) + (1/\xi)(V_C + V_K + V_{cl,C})]} \quad (8)$$

The Schmidt p - V diagram is presented in Fig. 10. Referring to Fig. 9; and applying the mass conservation equation among the spaces of the engine workspace [12];

For the expansion space;

$$dm_E/dt = 0.0 - m_{E \rightarrow H} \quad (9)$$

For the heater space;

$$dm_H/dt = m_{E \rightarrow H} - m_{H \rightarrow R} \quad (10)$$

For the regenerator space;

$$dm_R/dt = m_{H \rightarrow R} - m_{R \rightarrow K} \quad (11)$$

For the cooler space;

$$dm_K/dt = m_{R \rightarrow K} - m_{K \rightarrow C} \quad (12)$$

For the compression space;

$$dm_C/dt = m_{K \rightarrow C} - 0.0 \quad (13)$$

From the Eqs (9)–(13); one can find; $m_{E \rightarrow H}$, $m_{H \rightarrow R}$, $m_{R \rightarrow K}$ and $m_{K \rightarrow C}$. Hence the instantaneous flow rate and Reynolds numbers through the heater, the regenerator and the cooler were found as;

$$m_H = (m_{E \rightarrow H} + m_{H \rightarrow R}) / 2 \quad (14)$$

$$m_R = (m_{H \rightarrow R} + m_{R \rightarrow K}) / 2 \quad (15)$$

$$m_K = (m_{R \rightarrow K} + m_{K \rightarrow C}) / 2 \quad (16)$$

$$Re_H = m_H \times d_H / (\mu_E \times a_{H,min}) \quad (17)$$

$$Re_R = m_R \times d_{hyd,R} / (\mu_R \times a_R) \quad (18)$$

$$Re_K = m_K \times d_K / (\mu_C \times a_{K,min}) \quad (19)$$

Fig. 11 shows the cyclic mass fluctuations via the heater, the cooler and the regenerator during a complete cycle of the engine.

3. Heater

The flue gases flow inside the tubes of the elbow-bend heater, their temperature at inlet was taken 750 °C [13,14]. The flue gases flow rate was selected to be constant at 0.1378 kg/s to get $Re_g = 5000$.

Thus, the heat addition rate is [15];

$$\dot{Q}_H = N \oint p_E dV_E = A_{H,0} U_H (T_g - T_E) \quad (20)$$

Where;

$$1/U_H = 1/h_g A_i + 1/h_H A_o \quad (21)$$

The pressure drop through the heater is [16];

$$\Delta p_H = 1.6 \times [f \times \rho \times v_{max}^2 / 2]_H \quad (22)$$

The values of both h_H and f_H were calculated using the correlations (1) and (2). However the value of h_g was calculated from the following correlation [15];

$$\overline{Nu}_g = 0.023 Re_g^{0.8} Pr_g^{0.4} \quad (23)$$

4. Regenerator

The regenerator is a cylinder of 0.1 m diameter, as shown in Fig. 9. It is formed of successive circular layers of stainless steel wire-net having the same mesh size. The layers are supposed to be homogeneously stacked beside each other without a revolving angle. Mesh of $i = 100$ pores/in. and wire diameter of $d_w = 0.112$ mm was proposed for the regenerator fabrication. The pressure drop of the gas due to its fluctuation through the regenerator is [17];

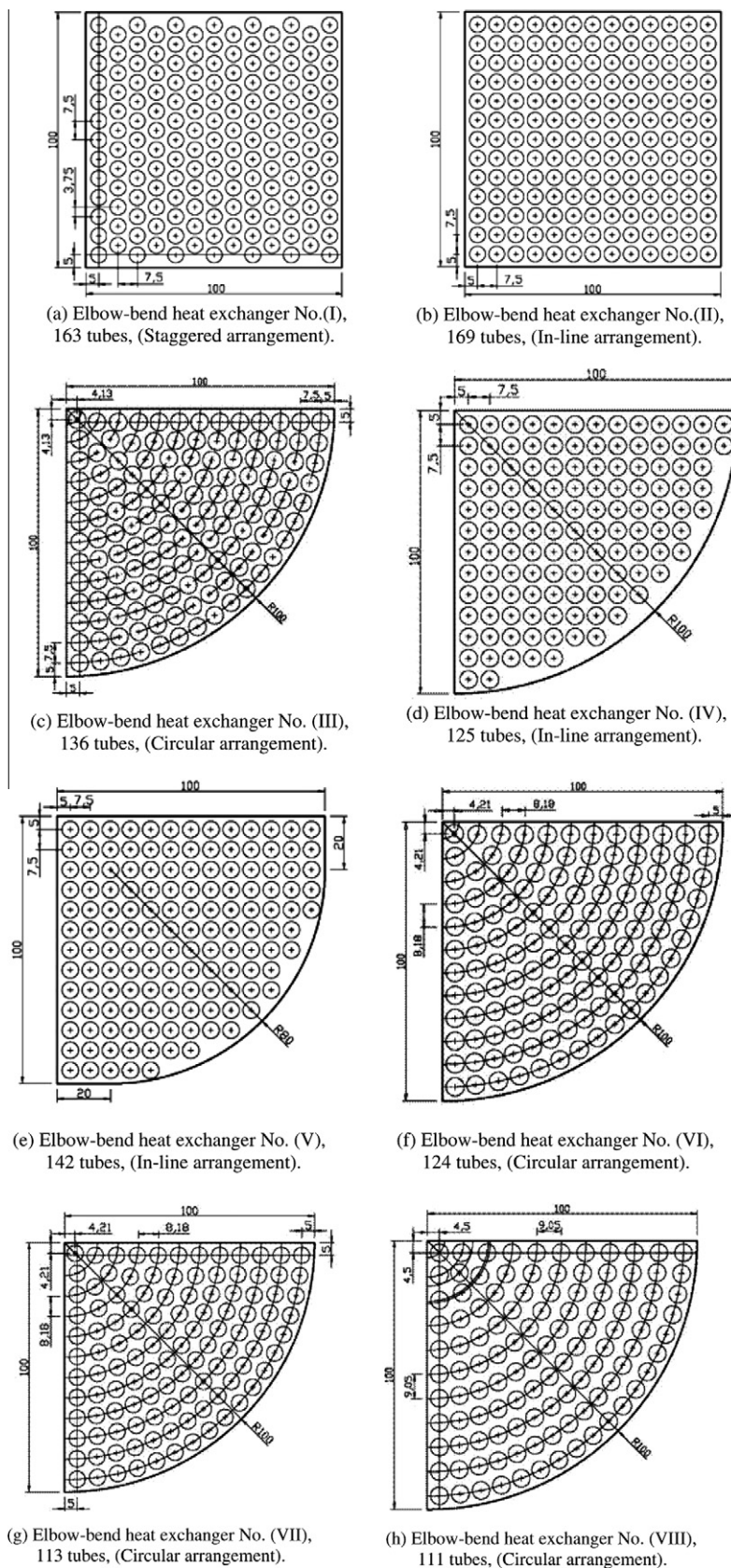


Fig. 8. Specifications of the eight elbow-bend heat exchangers.

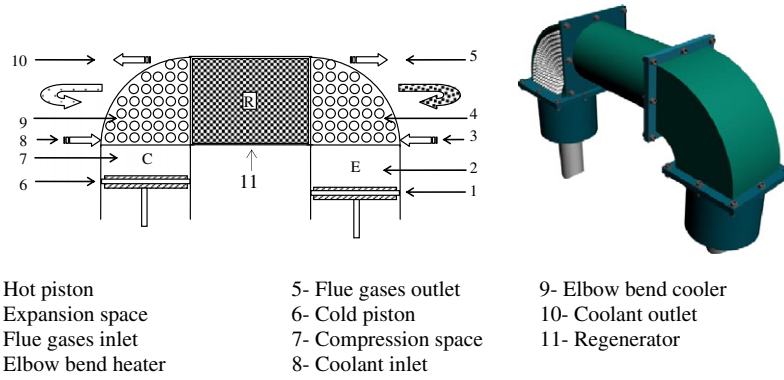


Fig. 9. Scheme of the work spaces of the proposed alpha-type Stirling engine.

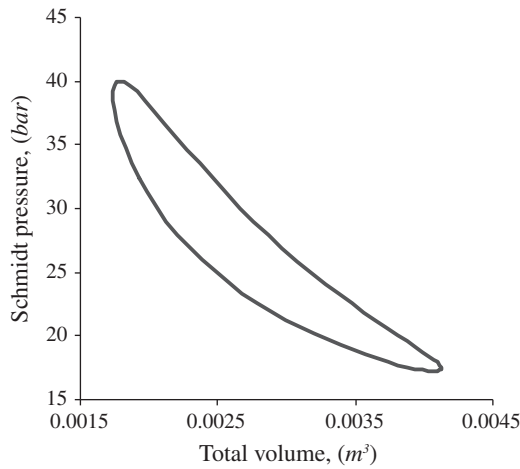


Fig. 10. Schmidt p - V diagram.

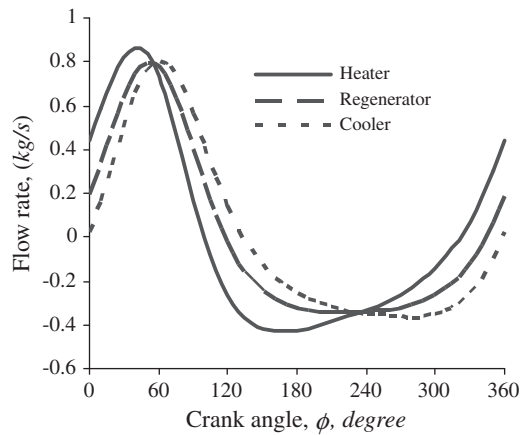


Fig. 11. Cyclic flow rate through heater, regenerator and cooler.

$$\Delta p_R = [f \times (L/d_{hyd}) \times \rho \times v^2/2]_R \quad \text{where } d_{hyd} = d_w \sqrt{\frac{\psi}{1-\psi}} \quad (24)$$

$$\psi = 1 - \frac{1000i}{25.4} \times \frac{\pi}{4} d_w \quad (25)$$

The friction factor, f , is expressed as follows:

$$\log(f_R) = 1.73 - 0.93 \log(Re_R) \quad \text{for } 0.0 < Re_R \leq 60 \quad (26)$$

$$\log(f_R) = 0.714 - 0.365 \log(Re_R) \quad \text{for } 60 < Re_R \leq 1000 \quad (27)$$

$$\log(f_R) = 0.015 - 0.125 \log(Re_R) \quad \text{for } Re_R > 1000 \quad (28)$$

The regenerator effectiveness is;

$$\varepsilon_R = NTU_R / (1 + NTU_R) \quad (29)$$

$$NTU_R = 2 \times \overline{St}_R \times L_R / d_{hyd,R} \quad (30)$$

$$\text{And } St_R = 0.595 / Re_R^{0.4} \times Pr_R \quad (31)$$

5. Cooler

The inlet temperature of the cooling water was assumed to be 40 °C. The water flow rate was considered to be constant at 0.3462 kg/s to get $Re_w = 5000$. The heat removal rate is [15];

$$\dot{Q}_K = N \oint p_c dV_c = A_{K,o} U_K (T_c - T_{wt}) \quad (32)$$

where

$$1/U_K = 1/h_{wt} A_i + 1/h_K A_o \quad (33)$$

The pressure drop through the cooler is [16];

$$\Delta p_K = 1.6 \times [f \times \rho \times v_{\max}^2/2]_K \quad (34)$$

The values of both h_K and f_K were calculated using the correlations (1) and (2). However the value of h_{wt} was calculated using the following correlation [15]:

$$\overline{Nu}_{wt} = 0.023 Re_{wt}^{0.8} Pr_{wt}^{0.3} \quad (35)$$

The hydraulic losses Δp_H , Δp_K and Δp_R were taken into consideration to get the actual pressure inside the expansion and compression spaces P_E and P_C , that in turn used for calculating power and efficiency.

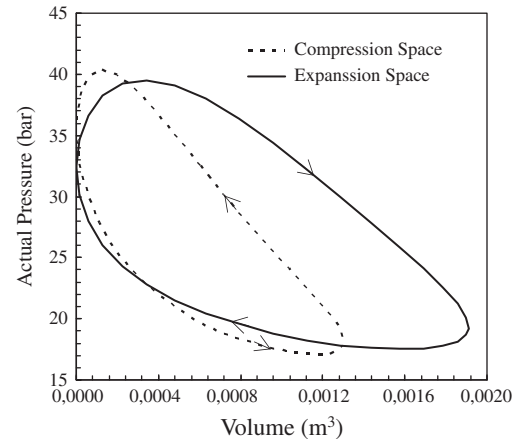


Fig. 12. p - V diagrams for both expansion and compression spaces.

Table 1

Maximum power developed for each elbow-bend heater and cooler for first design approach (non-equal strokes).

Heat exchanger no.	I	II	III	IV	V	VI	VII	VIII
L_R/D	0.505	0.503	0.503	0.505	0.505	0.506	0.506	0.506
S_E/D	2.404	2.437	1.998	1.989	2.117	2.096	2.148	2.337
S_C/D	1.6	1.66	1.5	1.5	1.55	1.5	1.50	1.55
α (deg)	96	86	112	106	106	110	110	106
N (rpm)	491	500	512	500	494	510	505	492
Power (kW)	8.175	8.729	7.456	7.270	7.592	7.507	7.533	8.161
η (%)	42.47	43.66	42.04	42.18	42.32	41.70	42.19	42.86
ϵ_R (%)	97.88	97.94	97.84	97.92	97.88	97.83	97.82	97.80
T_E (°C)	367.2	368.5	375.4	377.9	376.6	367.4	369.6	358.4
T_C (°C)	74.1	65.4	80.3	80.4	77.1	77.9	75.5	67.9
$T_E - T_C$ (°C)	293.1	303.1	295.1	297.5	299.5	289.5	294.1	290.5
m_{ch} (g)	54.04	51.05	47.79	48.25	50.68	50.56	52.98	54.31
p_{ch} (bar)	11.847	10.877	12.888	12.54	12.564	12.976	13.19	12.722
V_{dead} (cc)	862.254	905.023	509.817	639.37	662.799	598.498	663.393	688.25
V_{swept} (cc)	3144.73	3217.78	2747.32	2740.25	2880.06	2824.29	2865.13	3061.48
V_t (cc)	4006.99	4122.8	3257.14	3379.62	3542.85	3422.79	3528.53	3749.73
V_{dead} (%)	33.94	32.03	33.78	34.67	34.39	34.44	35.38	33.63
Specific power (W/cc)	2.6	2.71	2.71	2.65	2.64	2.66	2.63	2.67

Table 2

Maximum power developed for each elbow-bend heater and cooler for second design approach (equal strokes).

Heat exchanger no.	I	II	III	IV	V	VI	VII	VIII
L_R/D	0.505	0.503	0.503	0.505	0.505	0.506	0.506	0.506
S/D	2	2.06	1.77	1.77	1.85	1.815	1.842	1.957
α (deg)	96	86	109	106	102	107	106.5	101
N (rpm)	498	500	505	500	495	507	504	494
Power (kW)	8.098	8.650	7.408	7.127	7.515	7.443	7.478	8.091
η (%)	42.28	43.41	42.13	38.43	41.97	41.49	41.79	42.39
ϵ_R (%)	97.94	97.95	97.89	97.92	97.92	97.87	97.86	97.85
T_E (°C)	363.27	366.97	374.44	375.78	372.77	364.03	364.29	353.04
T_C (°C)	72.64	64.35	79.87	79.32	76.80	77.32	75.22	67.61
$T_E - T_C$ (°C)	290.63	302.62	294.57	296.47	295.97	286.71	289.07	285.44
m_{ch} (g)	5375	50.81	46.63	48.01	49.24	49.44	51.73	52.54
p_{ch} (bar)	11.859	10.833	12.430	12.413	12.064	12.586	12.767	12.223
V_{dead} (cc)	839.72	884.32	515.03	616.88	679.67	599.49	665.82	701.59
V_{swept} (cc)	3141.59	3235.84	2780.31	2780.31	2905.97	2851.00	2893.41	3074.05
V_t (cc)	3981.31	4120.16	3295.34	3397.19	3585.65	3450.48	3559.23	3775.64
V_{dead} (%)	34.16	32.05	33.39	34.49	33.98	34.16	35.07	33.40
Specific power (W/cc)	2.58	2.67	2.66	2.59	2.59	2.61	2.58	2.63

6. Power and efficiency

The elliptic p - V diagrams of the expansion and compression spaces when taking the hydraulic losses into consideration are shown in Fig. 12. The indicated power is [18,19]:

$$P = W_{net,cycle} \times N \quad (36)$$

$$W_{net,cycle} = (W_E + W_C)_{cycle} \quad (37)$$

$$W_{E,cycle} = \oint P_E dV_E = \sum_{i=1}^{i=n} (P_E \times \Delta V_E)_i \quad (38)$$

$$W_{C,cycle} = \oint P_C dV_C = \sum_{i=1}^{i=n} (P_C \times \Delta V_C)_i \quad (39)$$

The thermal efficiency of the engine is

$$\eta = P/Q_H \quad (40)$$

7. Results and discussion

Two different schemes for the engine were suggested to select its most suitable dimensions as ratios of the bore. The first consid-

ers non-equal strokes while the second one considers equal strokes. The experimental data of the heat exchangers that had

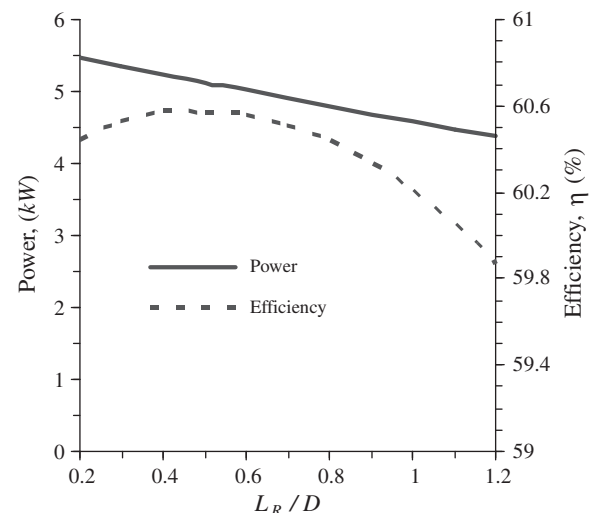


Fig. 13. Indicated power and efficiency versus the regenerator length for $S_E/D = S_C/D = 1.0$.

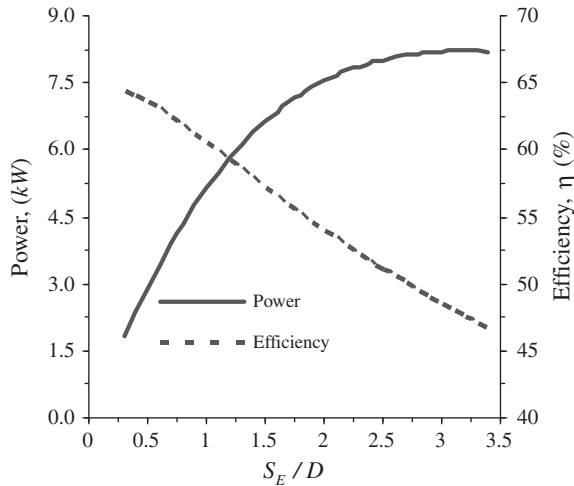


Fig. 14. Indicated power and efficiency versus hot piston stroke for $S_C/D = 1.0$ and $L_R/D = 0.503$.

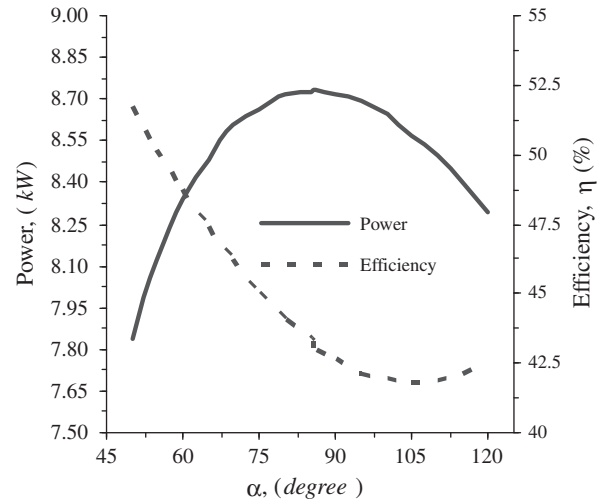


Fig. 16. Power and efficiency versus α for $\alpha = 86^\circ$, $S_E/D = 2.437$, $S_C/D = 1.66$ and $L_R/D = 0.503$.

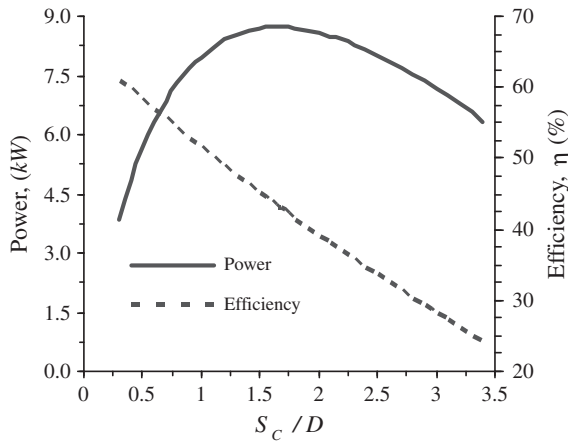


Fig. 15. Power and efficiency versus cold piston stroke for $S_E/D = 2.437$ and $L_R/D = 0.503$.

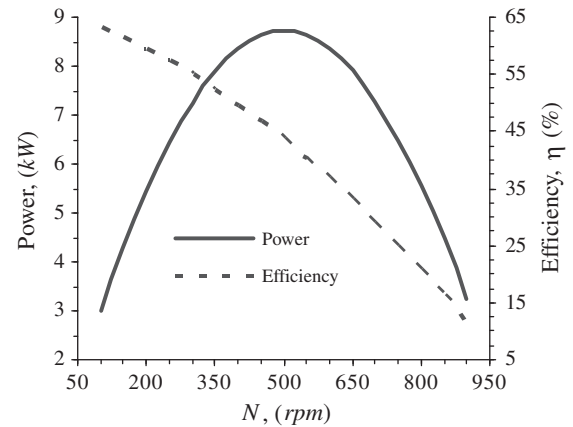


Fig. 17. Power and efficiency versus N for $\alpha = 86^\circ$, $S_E/D = 2.437$, $S_C/D = 1.66$ and $L_R/D = 0.503$.

been correlated in Eqs. (1) and (2) were fed to the computer program. The dimensions of the engine were varied to get the power and the efficiency. By trial and error method; the dimensions and conditions that lead to high power and efficiency were recorded. The results shown in Table 1 are for the first scheme and Table 2 is for the second one. Referring to Tables 1 and 2, one can say; the elbow-bend heat exchanger no. II is the most suitable candidate as a heater and a cooler for the two schemes. As a consequence; the results will be discussed when the heat exchanger no. II was used as a heater and a cooler in the proposed engine. The working fluid used in the engine is nitrogen and all the calculations were done for a maximum pressure limit of 40 bar. The increase in the regenerator length increases both the dead volume and hydraulic losses that in turn reduces the power. So, the most suitable length of the regenerator is about 0.0503 m as shown in Fig. 13.

In the typical Stirling engine design, the engine indicated power has a linear proportion with the swept volume in the vision of the classical Schmidt model without hydraulic losses. Furthermore, for the same dead volume, the indicated power increases with the increase of swept volume as the percentage of the dead volume comes down. This is more accentuated in Figs. 14 and 15 whereas; the increase in the hot piston stroke or cold piston stroke causes

the indicated power to go up although the present model attributes the hydraulic losses. For too long strokes, the charged mass of the working fluid increases and as a consequence the velocity through the heat exchangers increases which increases the hydraulic losses. As a consequence, the indicated power comes down.

The variation of the phase angle varies the maximum and minimum total volume of the engine work space which causes a variation in the compression ratio. Fig. 16 shows that the optimum phase angle between the hot and cold pistons of the proposed engine is about 86° . It can be noted that the indicated power increases with the engine speed up to a certain limit as the number of cycles per unit time increases. At higher speeds, the indicated power decreases with increasing the engine speed due to the hydraulic losses and the inadequate heat transfer at higher speed levels. For the proposed engine; the optimum speed is around 500 rpm as shown in Fig. 17.

8. Comparison among the present work and literature works

An alpha engine of 1 kW target with a heat source of 300°C was tested experimentally by [20] at different engine speeds. The en-

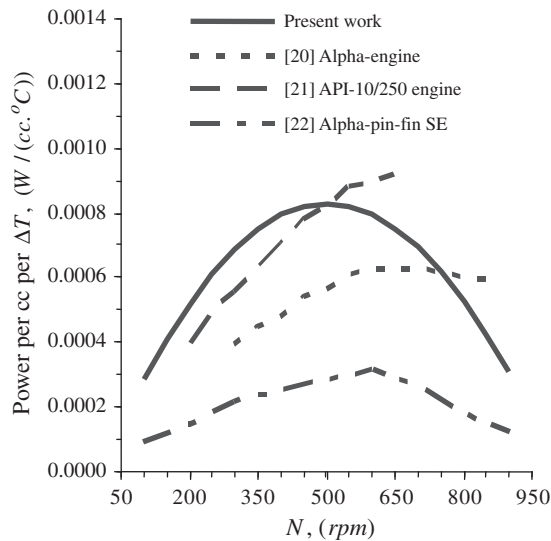


Fig. 18. Comparison among the power per cc per ΔT of the present work and those by the literature works.

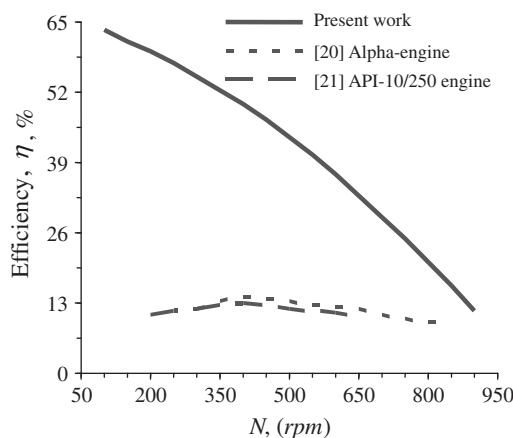


Fig. 19. Comparison of the efficiency of the present work and those by the literature works.

engine has a compression ratio of 1.23 and it developed a power of 805 W at 700 rpm. A prototype alpha-Stirling engine API-10/250 used a heat source of 300 °C and a heat sink of 20 °C was tested experimentally at different engine speeds by [21]. It generated a power of 10.4 kW. An alpha engine with a heat source of 237 °C and a heat sink of 30 °C was tested experimentally by [22] at different speeds and different values of temperature difference. The experimental tests of this engine showed that the relation between the maximum power and the temperature difference is almost linear.

Fig. 18 compares the indicated power per cc per unit temperature differential between the heat source and the heat sink versus engine speed of the present engine with the corresponding actual power of the engines of [20–22]. A quite trend agreement among the present results and those by the literature was more attained. In low speed range, the present engine delivers about 13% higher indicated power per cc per ΔT than that by the literatures. This is due to the higher temperature differential of the present engine than that of the three engines which causes adequate heat transfer rates at low engine speeds. However, at high speed range; the inadequate heat transfer rates cause the power to come down. As the present engine has higher ΔT than that of the engines in compar-

ison, the present engine operates at higher efficiency than that of the others as shown in Fig. 19.

9. Conclusions

The present work concerns with a somewhat new innovation of an alpha-Stirling engine using elbow-bend heat exchangers as a heater and a cooler. The experimental results of eight elbow-bend heat exchangers were used in the design of the engine. The engine has twin cylinders on a common crankcase and it uses nitrogen as a working fluid at a maximum operating pressure of 40 bars. The elbow-bend heat exchangers were introduced as a heater and a cooler in the proposed engine. The main dimensions of the engine that result in higher power were found out in the vision of Schmidt theory. The results of the proposed engine were compared with the results of three practical engines. The concluded items of the present work are summarized in the following items:

1. The elbow-bend heat exchangers having straight tubes are easy to manufacture, have long life time, reliable, light weight and quite cheap to be suitable candidates in Stirling machines compared with the conventional heat exchangers having curved tubes.
2. Elbow-bend heat exchangers reduce the hydraulic losses but they slightly increase the dead volume. The net result is an increase in the power by about 13% at high efficiency levels.
3. Upon calculation, a target engine is rated at 9 kW/400:700 rpm/4000 cc, with nitrogen as a working gas could be achieved.
4. Even though, the elbow-bend heat exchangers can be suggested for gamma, double-acting and free-piston Stirling machines. Moreover, they can be used in other thermal applications.

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